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Original Article

NATURAL FREQUENCY RESPONSE OF ROTOR SHAFT TO CRACK DEPTH AND CRACK LOCATION

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ABSTRACT

Rotating components of various machines as turbine, pumps, generators and compressor may fail due to crack generation and propagation. Rotor shaft is considered as one of the important part of various rotating machines. Due to manufacturing defect or cyclic loading, fatigue crack may appear in rotating shaft. Crack is considered as one of the main reasons for catastrophic failures in rotating shaft. The diagnosis of shaft faults has been gaining importance in recent years. To examine the vibration characteristics of cracked shaft, a steel shaft supported by two bearings was used. The shaft models with various crack locations and crack depth were designed in CATIA. Different crack depths 0.16, 0.33, 0.5, 0.66, 0.83 & 1 percent of maximum crack depth were provided to shaft model. Crack locations altered 75, 150, 225, 300, 375, 450 mm from bearing support. Modal analysis of these shaft models carried out using ANSYS. Natural frequencies obtained from modal analysis were analysed. Natural frequencies obtained from modal analysis were validated with whirling of shaft experimental set up.

KEYWORDS: Fatigue crack, Artificial Crack, ANSYS, Modal Analysis & Crack Location

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1. INTRODUCTION

Crack detection and localisation is gaining interest of reasearchers, across the globe. Modal analysis is important tool to observe crack sevearity. M. J. Gomez et al. observed wavelet packet transform energy by varying crack depth. Wavelet packet transform energies used to identify crack depth with help of artificial neural network [1]. Tejas Aher et al. obtained natural frequency of shaft from FEA. Crack severity predicted using natural frequency of shaft [2]. L Deng and R Zhao used vibration signal for condition monitoring of rotor bearing system. Local mean deformation technique and Fourier transform used for vibration analysis [3]. B James Prasad Rao et al. carried out failure analysis of composite shaft using maximum stress criteria [4]. D siano et al carried out vibration based spectral analysis of pump using FFT [5]. V Sudheer Kumar et al. observed change in displacement curve as crack depth increases [6]. Author prepared finite element model of composite shaft. Natural frequency and critical speed obtained from finite element modal by changing stacking sequence and fiber orientation [7], [8]. Alzbeta Sapietova and Vladimir Dekys analyzed misalignment of shaft using frequency spectram analysis [9]. A. Teter and J. Gauryluk carried out experimental modal analysis of composite blade with LMS analyzer [10]. M. T. Das and Ays Yilmaz observed that as crack depth increases, natural frequency of vibration decreases [11]. J. Yao et al. carried out harmonic analysis of rotor shaft using FFT analyzer. New approach for harmonic extraction proposed using

discrete Fourier transform [12]. H. P. Phadtare and Barun Pratiher observed resonance behavior of shaft bearing system at high speed [13]. S. Ben Arab et al. observed variation in critical speed and natural frequency due to change of stacking sequence and fiber orientation [14]. De Paula Mendonca et al. carried out vibration analysis of rotor mounted on composite shaft [15]. P. Borghesani et al. proposed velocity synchronous Fourier transform to analyze high speed rotary machine components [16]. Mutra Rajasekhara Reddy and J. Srinivas modeled shaft rotor system using FEA [17]. C. Y. Chang et al. analyzed vibration properties of composite shaft with randomly oriented reinforcement. It was observed that dynamic properties as natural frequency and critical speed get altered due to orientation of reinforcement [18]. A. Khadersab et al. used FFT spectrum technique to identify bearing faults. It was observed that FFT spectrum technique can be effectively used to determine bearing faults [19]. Almuslmani M. and Ganeshan R. formulated FEA model of tapered composite shaft. Variation in critical speed and natural frequency observed with respect to stacking sequence and tensile load on shaft [20]. Victor Girondin et al. carried out vibration analysis of helicopter shaft spectral coherence clearly indicate propagation of crack [21]. P. Satheesh Kumar Reddy and Nagaraju carried out weight optimization of composite drive shaft [22]. Sudhakar I. et al. carried out condition monitoring of 3 phase induction motor using FFT. Bearing cracks and misalignments of shaft detected during experiment [23]. M. Eishamy et al. observed natural frequency of cantilever beam by changing crack depth and crack location [24]. C. Elanchezhian et al. carried out comparison of steel drive shaft with composite drive shaft. Weight and strength were criteria of comparison [25]. A. P. Stawiarski [26] found the effectiveness of vibration based crack detection depends on a comparison of relative change of shaft compliance. J. Xiang et al.[27] developed a method for crack detection using wavelet transform and genetic algorithm. Victor Girondin et al. [28] developed a mathematical model for the healthy and cracked shaft. X Li et al. [29] developed a maintenance policy for gear shaft assembly and modelled deterioration of shaft and gear. Isauflas et al. [30] proposed a health monitoring system for the transmission shaft. B. Eftekharnejad et al. [31] observed that crack diagnosis of transmission shaft can be carried out efficiently by a combination of acoustic emission, vibration and motor current analysis method. M. Karthikeyan et al. [32] found that crack depth and location can be identified by natural frequency and forced response analysis. R Citarella and G Cricri [33] investigated crack growth behaviour using dual boundary element method and FEM. T Szolc et al. [34] carried out damage identification in rotary shaft using Monte Carlo sampling approach. S. S. Naik and S. K. Maiti [35] carried out a vibration analysis of Timoshenko and Euler Bernaulli shaft with crack arbitrarily located. P Pennacchi et al. [36] carried out model-based crack identification in the frequency domain. Z. N. Hajis and Olutunde Oyadiji [37] used orthogonal natural frequencies to identify and locate the crack. L Rubio et al. [38] identified the crack location using natural frequency and anti-resonant frequency data. C. Hour and Y Lu [39] developed a finite element model to detect a crack in a thick beam. Deviation in natural frequency occur due to change of crack depth and crack location. In the present study, new approach developed to study variation in crack depth and crack location. Critical speed range should be avoided to avoid failure of shaft due to resonance. If natural frequency of shaft becomes equal to frequency of rotary shaft, resonance will occur. Therefore, study of natural frequency of shaft by changing crack location and crack depth carried out in this experiment.

2. EXPERIMENTAL PROCEDURE

As shown in figure 1, shaft supported between two bearing and rotated with help of electric motor. Crack of depth 0.16, 0.33, 0.5, 0.66, 0.83 & 1 percent of maximum crack depth were provided to shaft model. A single crack was provided at various locations 75, 150, 225, 300, 375, 450 mm from bearing support. Shaft rotated with help of electric motor.

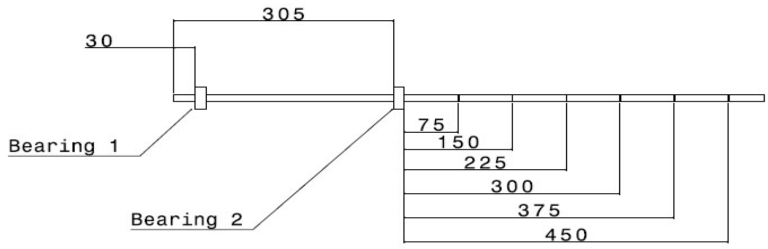

Figure 1: Crack Locations from Bearing Support.

Figure 2 shows experimental set up for whirling of shaft. Electric motor rotates shaft with help of pulley mounted on it. Variable voltage drive alters voltage of electric motor. Speed of motor varies due to change in voltage

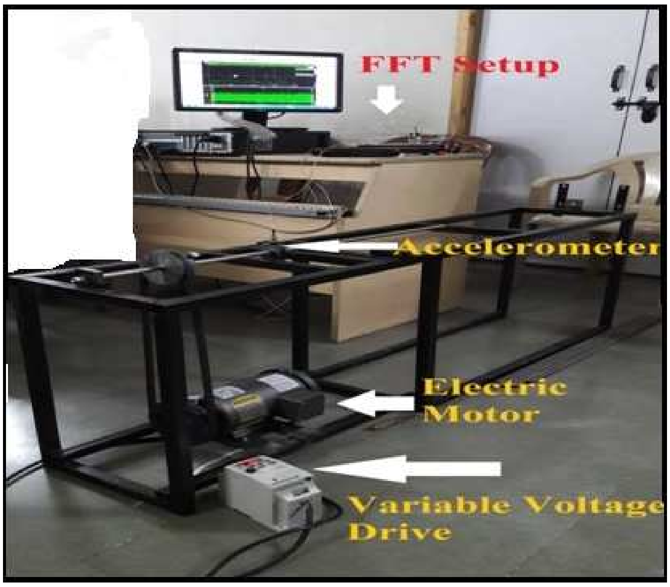

Figure 2: Experimental Setup.

Following parts were used to make experimental set up.

Shaft

A shaft of 880mm length and 12mm diameter used for experiments.

Electric Motor

A three phase Electric Motor rotate the shaft with help of v belt and pulley mounted on it. The motor was 1/3 H. P. and 0.25 K. W. power.

Variable Voltage Drive

Variable voltage drive used to vary speed of shaft. Variable voltage drive works on 3-phase power at 208 V to240 V. The speed of the electric motor can be altered by using variable voltage drive.

Tachometer

Tachometer was used to measure rotations, per minute of shaft. Tachometer probe applied on rotary shaft to get RPM reading.

Pulley and Belt

One pair of pulley used in setup. One pulley mounted on motor shaft and another on shaft. A v-type belt used in setup to power transmission of motor to shaft.

Bearings

Figure 3 shows shaft, supported in bearing. The bearing used in this experiment was single row deep groove ball bearing that can take radial load and thrust load. The bearing shouldered in housing, made of Mild steel with press feeting to secure adequate support for the bearing and resist the maximum thrust load. The press fitted plates were welded by electric welding to the frame, in proper alignment.

Figure 3: Bearing and Bearing Housing.

Shaft with single crack rotated from steady position and gradually speed of shaft increased. When shaft vibrates with maximum amplitude, speed of shaft noted. This procedure repeated for all crack depth and crack locations. Speed of shaft converted to frequency of vibration. Experimental procedure mentioned below.

The main objective of experiment was to get natural frequency values, by varying crack depth and crack location. Each type of shaft was supported between two bearings. Electric motor rotate the shaft trough pulley and belt arrangement. A variable voltage drive used to vary speed of electric motor. Speed of electric motor increased gradually from steady position to maximum speed limit.

The shaft with crack location 75 mm and crack depth 0.16 percentage of maximum crack depth supported between two bearings. Electric motor used to rotate shaft. As rotary shaft speed reaches to its critical speed, it start to vibrate with maximum amplitude. At Critical speed natural frequency of shaft and frequency of rotary shaft becomes equal. With maximum amplitude, rotary shaft vibrates when resonance occur. Speed of shaft measured with tachometer. Frequency of vibration of shaft can be calculated with help of critical speed value.

As we get value of critical speed for first crack depth, electric motor stopped for some time. Depth of crack increased from 16 percent to 33percent of maximum crack depth. To avoid misalignment error, depth of crack increased in a loaded condition. Procedure mentioned in above paragraph repeated to tachometer reading of critical speed value. Same procedure repeated for crack depths0.5, 0.66, 0.83 & 1 percent of maximum crack depth. Tachometer readings were noted for each crack depth. After taking all crack depth readings, shaft removed from experimental set up. New shaft for crack

location 150mm used. New shaft supported in bearongs. Crack of depth 0.16, 0.33, 0.5, 0.66, 0.83 & 1 percent of maximum crack depth were given to new shaft and tacho meter readings of critical speed were noted. Same procedure repeated for shafts with crack location 225, 300, 375 and 450mm.

3. MODAL ANALYSIS

After experimentation, these experimental results checked using ANSYS software. This validation consists of finite element analysis of cracked shaft at different values of crack depth. Analysis has been carried out in ANSYS package. Modelling of healthy and cracked shaft in FEM discussed. Material for shaft was steel 304. Material properties of steel 304 are as shown in table 1.

Table 1: Material Properties of Shaft

Material	Density	Young's Modulus	Poison's Ratio
Steel 304	8000Kg/m3	193E+09	0.29

Cad Modelling

Diameter of shaft model was 12 mm. Maximum crack depth taken as 6 mm. modelling of cracked shaft carried out in CATIA V5. A single crack developed at different crack depth 0.16, 0.33, 0.5, 0.66, 0.83 & 1 percent of maximum crack depth generated. Crack location altered 75, 150, 225, 300, 375, 450 mm from bearing support. Total 36 models of single cracked shaft generated using six crack depths, at six crack locations. Figure 4 shows cad model of shaft.

Figure 4: Cracked Shaft Model.

Meshing

The solid model is imported to Ansys 16, and mesh generated for the same. Boundary conditions are applied to meshed model. The element has three degrees of freedom at each node translations in the nodal x, y, and z directions. Size of mesh element was 0.03mm. The bearing outer case was fixed in all degrees of freedom. The number of elements created were 1, 65,238. Figure 5 shows meshed model of shaft.

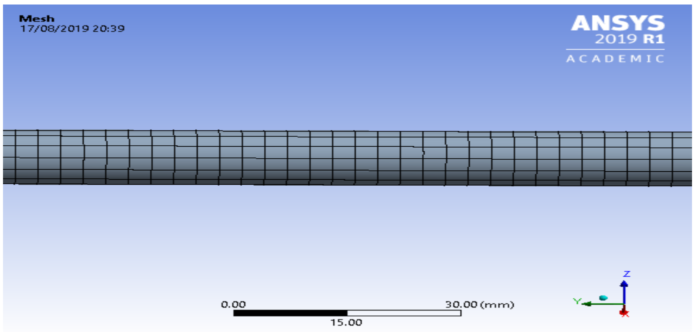

Figure 5: Meshing of Shaft.

Boundary Condition

Meshed shaft model supported in bearing. All degree of freedom of bearings were fixed. Modal analysis performed on shaft model and first three natural frequencies extracted. Model shaped vibration of the rotor shaft obtained.

4. RESULTS AND DISCUSSIONS

First, three mode shapes observed and natural frequencies extracted. Natural frequencies analysed by changing crack depth. Effect of crack location on natural frequencies observed. Section 4.1 discusses mode shapes of shaft and bearing system. Section 4.1 describes mode shapes obtained from nodal analysis. Section 4.2 discusses Effect of crack location on natural frequency. Section 4.3 describes Effect of crack depth variation on natural frequency.

Modal Analysis

Mode shapes of shaft and bearing system observed. Figure 6 shows first mode of vibration. Second mode of vibration shown in figure 7. Figure 8 depicts third mode of vibration.

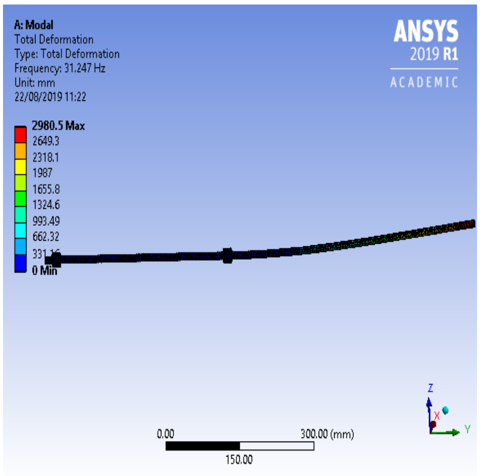

Figure 6: First Mode of Vibration.

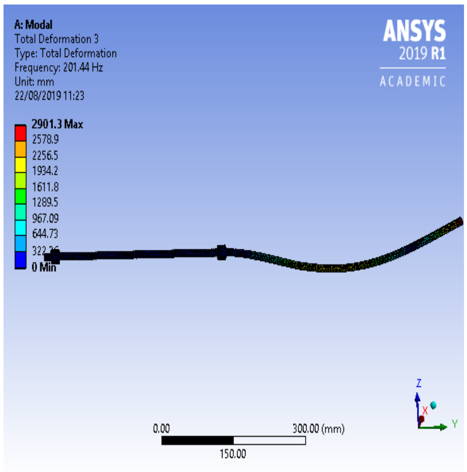

Figure 7: Second Mode of Vibration.

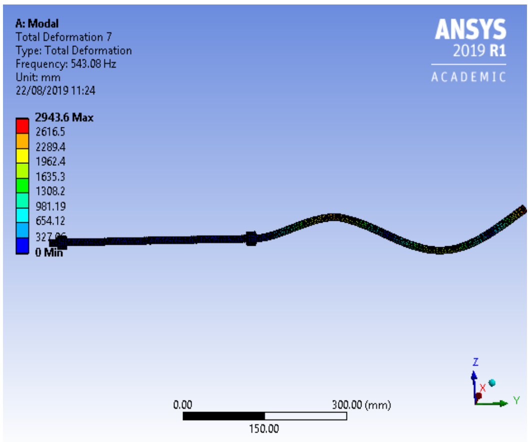

Figure 8: Third Mode of Vibration.

Effect of Crack Location on Natural Frequency

On shaft, a single crack was generated at various location (75, 150, 225, 300, 375 and 450 mm from bearing support) with varying depth (0.16, 0.33, 0.5, 0.66, 0.83 & 1 percent of maximum crack depth). Effect of variation in crack location on natural frequency was observed. Figure 9 shows natural frequency versus crack location graph for first mode of vibration. Figure 10 represents natural frequency versus crack location graph for second mode of vibration. Figure 11 depicts natural frequency versus crack location graph for third mode of vibration.

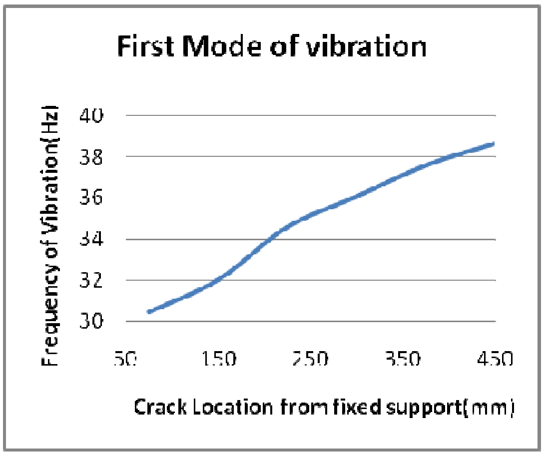

Figure 9: Frequency Vs. Crack Location for First Mode of Vibration with 2mm crack Depth.

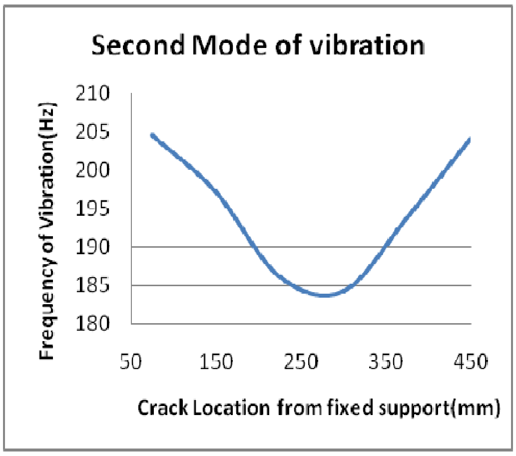

Figure 10: Frequency Vs. Crack Location for second Mode of Vibration with 2mm crack Depth.

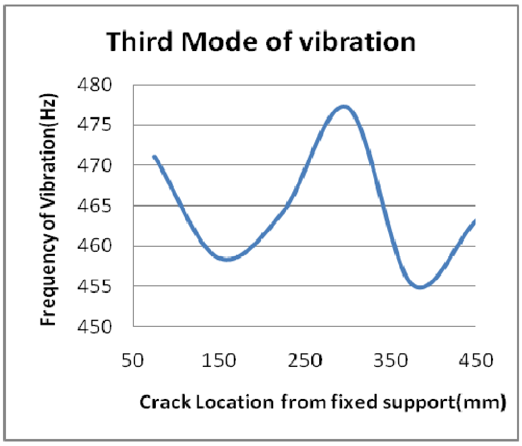

Figure 11: Frequency Vs. Crack Location for Third Mode of Vibration with 2mm crack Depth.

Effect of Crack Depth variation on Natural Frequency

Figure no 12, 13 and 14 represent frequency vs relative crack depth for first, second and third mode shaped, respectively. Crack depth variation effect on natural frequency was observed. It was observed that frequency of vibration decreases with crack depth.

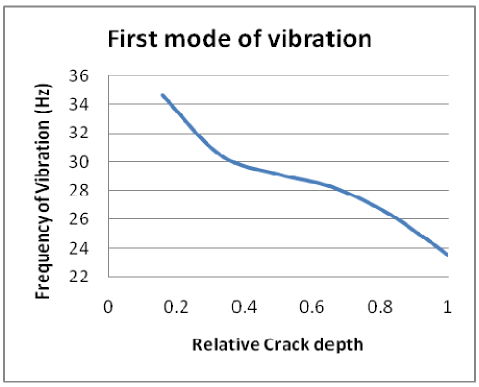

Figure 12: Frequency VS. Relative Crack Depth for First Mode of Vibration with 2mm Crack Depth.

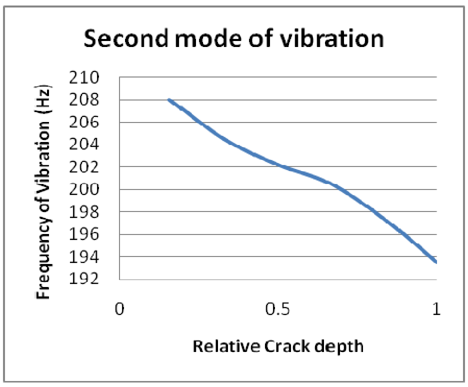

Figure 13: Frequency Vs. relative crack Depth for Second Mode of Vibration with 2mm Crack Depth.

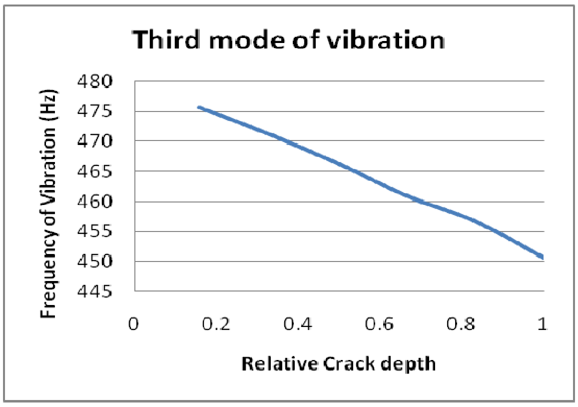

Figure 14: Frequency Vs. Relative Crack Depth for Third Mode of Vibration.

5. CONCLUSIONS

In the present experiment, models of shaft with different crack location and crack depths were prepared. Modal analysis of all shaft models were carried out to obtain the frequency of vibration. First three mode shapes of shaft with crack were observed. First three natural frequencies of vibration obtained from modal analysis. Experimental set up of shaft bearing system prepared. Natural frequencies obtained from modal analysis, validated with help of experimental set up.

These natural frequencies analised for varing crack depth. Natural frequencies also analised for change of crack location. It is observed that as crack depth increases, frequency of vibration decreases for first, second and third mode of vibration. Natural frequency of vibration increases as distance of crack from bearing support increases for first mode shape. Second and third mode of vibration graph of frequency vs crack depth was not linear.

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